Large Environmental Chamber: Design and Operating Characteristics

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Abstract. *Environmental chambers are important tools for probing the influence of the environment on biological systems. This article describes the design and operating characteristics of a large environmental chamber (7.4 × 10.5 × 3.4 m). The impetus for constructing the chamber was to investigate how environmental conditions affect gaseous emissions, including ammonia and odorous compounds, from animal housing facilities and from manure processing technologies. The chamber is currently configured to house six dairy cows in tie–stalls. Mass air flow for the chamber can be set in the range of 5.6 to 28.8 air exchanges per hour and, with some limitations, can be measured and controlled to within 1% of the setpoint. Exhaust air temperature can be controlled to within 0.2°C of the setpoint. Light intensity can be set to simulate diurnal variations due to solar radiation. Discussed is the chamber design and operating characteristics, how to avoid costly mistakes, and tradeoffs that can be made to decrease construction and operating costs.*

Keywords. *Environmental chamber, Ammonia emissions, Livestock buildings, Dairy cattle.*

Environmental chambers are commonly used to study the impact of the environment on plants and animals. The choice of environmental variables to control along with control tolerances depends on experimental objectives and implementation costs. The most commonly controlled variables are ambient temperature, humidity, and air flow. Temperature is important because rates of most chemical, and therefore biological, processes are temperature dependent (Langhans and Tibbitts, 1997). Humidity can be important as it affects rates of heat transfer and respiration, particularly in plants (Langhans and Tibbitts, 1997). Air velocity and exchange rates can influence surface exchange phenomena such as drying and heat loss, and can be of critical importance in cases where gas fluxes must be measured; e.g., in the case of indirect calorimetry (Brown–Brandl et al., 1998; Neinaber and Maddy, 1985). Lighting can also be an important variable as light intensity and duration effect plants (Lumsden and Millar, 1998) and animals (Turek and Van Reeth, 1996). Designing chambers involves consideration of construction and operating costs, as well as accuracies and dynamic characteristics that controlled variables must satisfy.

This article describes the design and operating characteristics of a large environmental chamber. The impetus for constructing the chamber was to investigate how environmental conditions affect gaseous emissions, including ammonia and odorous compounds, from animal housing facilities and from manure processing technologies such as composting. The two environmental factors with the greatest impact on these emissions are temperature and air velocity (Elzing and Monteny, 1997). In contrast, humidity is not a factor in most models of emissions as models are founded on theoretical gas exchange equations (Anderson, 1995; Monteny et al., 1998). However, humidity can affect the surface characteristics of manure. When the chamber was tested with six cows in the chamber, ammonia volatilization rates were reduced 10–30% when the relative humidity fell below 20%; presumably due to rapid crust formation on the manure which reduced volatilization (personnal observation). Limited practical design information was available at the time the chamber was planned which resulted in additional accrued expenses and construction time delays. Discussed is the chamber design and operating characteristics, how to avoid costly mistakes, and tradeoffs that can reduce construction, and operating costs.

Materials and Methods

Overview

Chamber Layout

An overview of the chamber infrastructure is shown in figure 1. The internal dimensions of the chamber are 7.4 m wide × 10.5 m long × 3.4 m high and it contains approximately 260 m^3 of air space. The chamber is made of a prefabricated aluminum panels, which are 10.2 cm (4 in.) thick and contain injected foam insulation. The panels are hung from a support structure attached to the ceiling. Walls at floor level are stabilized using I–beam girders sunk halfway into a cement floor; the panels fit within the recess of the I–beam and are caulked in place (Nor–lake, Hudson, Wis.). For one wall, which is accessible to the animals, 1.9–cm (.75–in.) pressure–treated plywood was inserted

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behind the aluminum sheeting for the inside wall before the foam was injected. Electrical conduits are contained within the walls. Air enters the chamber through six 41– × 41–cm (16– × 16–in.) ceiling ducts that run longitudinally along the edge of the chamber, and exits across the chamber through six 41– × 41–cm (16– × 16–in.) wall ducts centered 46 cm (18 in.) above the floor. Six 41– × 25–cm (16– × 10–in.) vertical ducts channel exhaust air to a single large plenum above the chamber. This rise was designed to trap large particulates. Louvers in the ceiling and exhaust ducts allow air flows to be balanced. Supply and exhaust ducts are centered relative to the six animal tie–stalls to allow for the possibility of subdividing the chamber. The main supply and exhaust plenums are 102– × 36–cm (40– × 14–in.). There are four 30– × 30–cm observation ports.

Air Flow

To allow for separation of the air intake and exhaust, the intake is at ground level outside the building that houses the environmental chamber and the exhaust is on the roof of the building. Air is discharged horizontally in the opposite direction of the intake. The intake air path is through a filter bank, the main cooling coil, the supply fan, an electric heater, and then a steam coil before it reaches the chamber. The electric heater was added to allow for the possibility of using the chamber at below freezing temperatures when outside ambient temperatures remained below the desired chamber temperature. A humidifier was not added as ambient humidity in the warmer months is normally high and, during winter, animals were expected to contribute sufficient moisture to maintain comfortable humidity levels. The exhaust air path includes a 4.6–m section of 46–cm (18–in.) diameter circular duct incorporating an air flow measuring station and it ends with the exhaust fan housing.

Cooling

The cold water cooling system is outlined in figure 2. When conditions warrant, cold water from the chiller is pumped through the main cooling coil. A three–way bypass valve is used to control how much of the circulating water goes through and how much bypasses the coil. The figure shows an auxiliary cooling coil located in the chamber. After the chamber was built, this air handler was removed from the chamber (see Results and Discussion).

Animal Housing

The chamber contains six stainless–steel tie–stalls for dairy cows with rubber comfort mats. The stalls are at ground level and manure is collected using grates that are located behind that the stalls. The grates cover a recessed scrapper system. The floor is made of cement and bedding can be used if desired. The stalls are bolted in–place and can be removed or reconfigured.

Specifications

Controller

The digital controller is housed in an National Electrical Manufacturers Association (NEMA) Type 4 enclosure, which has space for 24 modules (System 600 Modular Building Controller, Siemens Building Technology, Beltsville, Md.). The controller cycles through the control program code approximately every second. Inputs and outputs are updated as required by programming. Values of variables used in the control program can be displayed in real–time, or stored by time or in response to error conditions. Data for each selected variable can be stored at 1–min intervals or at multiples thereof. As configured, the controller contains two processor units each with 2 MB of memory

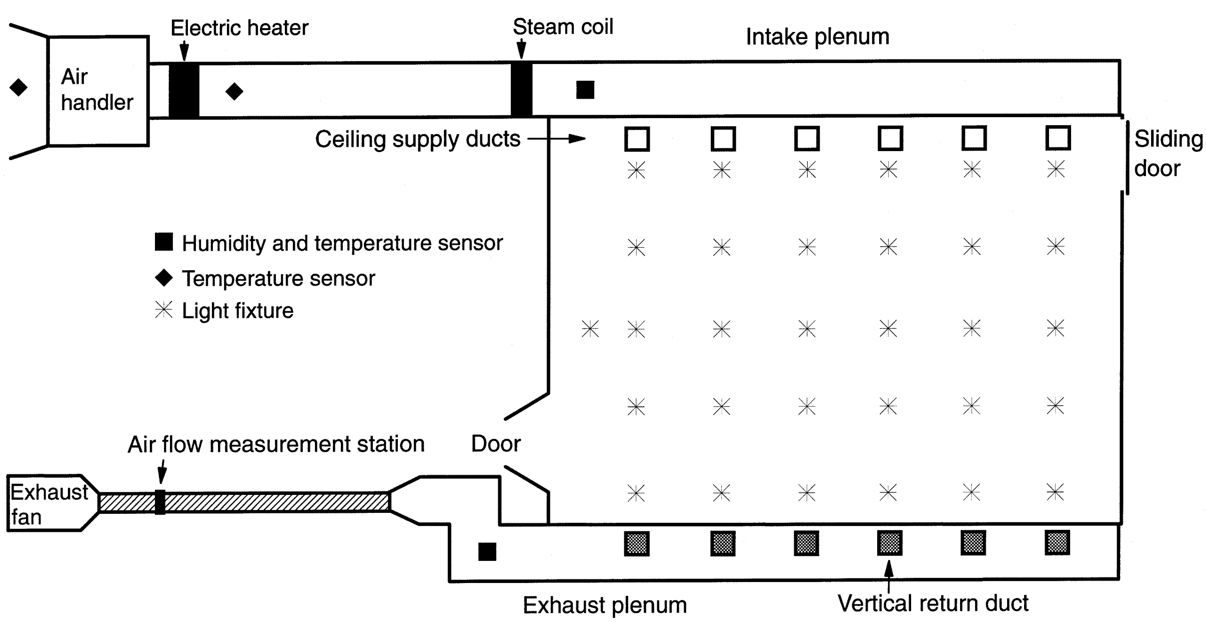

Figure 1. Top view of an outline of the infrastructure for the environmental chamber. The air handler is on the floor of the building and the electric heater is actually in the ductwork between the air handler and the ceiling mounted intake plenum. The air handler contains a filter bank, the supply fan, and the main cooling coil. The intake and exhaust plenums are attached to the building ceiling and are above the height of the chamber. The exhaust fan is mounted in the roof of the building. Connections between the ceiling air vents and the supply duct and between the exhaust air vents and vertical return ducts are not shown.

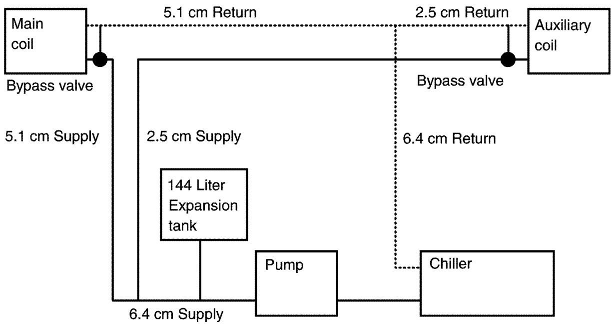

Figure 2. Overview of the cold water cooling system.

along with a power supply unit, which together occupy six module slots. Three dual 0– to 10–V modules, one dual 0–to 20–mA module, and four dual contact closure modules provide outputs. Two modules each with four contact closures, five dual 0–20 mA modules, and one dual 0– to 10–V module provide inputs. Analog resolution is 12–bits. Serial ports on the control modules are connected to a printer and a modem. An adjacent enclosure houses power supplies, an uninterrupable power supply, and an automated telephone dialer (see Alarms below; Model AD–2001, United Security Products, Inc., San Diego, Calif.). The modem is connected to the phone dialer, which, in turn, is connected to a dedicated phone line. The controller can be programmed locally using a portable computer running a terminal emulation program or remotely by calling the modem.

Supply and Exhaust Fans

Both fans are 3 hp, 208 V, 3–phase, and 60 cycle (High Efficient E+3, Trane Commercial Systems, La Crosse, Wis.). The variable frequency controllers have 0– to 10–V inputs and allow 6–to–1 air–flow turn–down ratios (Model GPD505, Magnetek, Inc., Nashville, Tenn.). The chamber was originally designed to allow volumetric air flows of 23 to 136 m^3/min (800 to 4800 cfm).

Cooling

The cold water chiller is nominally rated at 20 tons, incorporates hot gas bypass, and is set to discharge cold water coolant at 2°C (Model CGAF–C20, Trane Commercial Systems). A 50:50 mixture of propylene glycol and water is used as the coolant. The main cooling coil uses four row cooling with aluminum fins and 1.3–cm (0.5–in.) diameter tubing, and was specified for an air flow of up to 136 m^3/min.

Cold Water Pump

The cold–water pump capacity is 197 L/min (52 gpm) at a developed head of 265 kPa (38.5 psi). A relay activated by current flow to the pump is used to assure that the pump is operating after it is activated (Hawkeye, Veris Industries, Inc., Portland, Oreg.).

Electric Heater

The electric heater has six elements that can be individually activated (Redd–i, Gray, Tenn.). Using a sequencer (Model UCS–621, The Kele Companies, Bartlett, Tenn.), a 0–to–20 mA signal from the controller is used to activate six relays and the associated heating elements, in a step–wise manner. This layout allows control of a discrete device such as the heater with a continuos output signal. Alternatively, the selected number of elements can be activated using the appropriate mA output signal.

Steam Coil

The steam coil is a non–freeze model with 2.3 aluminum fins per cm. The finned area is 61 × 91 cm (24 × 36 in.). The tubing is made of copper and is 2.5 cm (1 in.) in diameter.

Control Valves

The valve regulating steam flow and the three–way bypass valve regulating flow of cold water to the cooling coil are controlled using 0– to 10–V actuators (Flowrite; Siemens Building Technology). A zero volt signal corresponds to no steam flow and 100% water flow to the respective coils.

Door Opening

Magnetic switches are used to monitor when either chamber door is open.

Lighting

To allow the chamber to be subdivided and light intensity to be programmed on a time basis, five incandescent fixtures are equally spaced between each pair of supply and exhaust ducts (fig. 1). One additional fixture is located near one door. The 31 fixtures are waterproof and are rated for 100–watt incandescent bulbs. To increase the life of light bulbs, light output is reduced to 95% of capacity for a total heat load of 2945 W. This wattage requires the use of two electric circuits. Each circuit is attached to a variable dimmer (Model HP–2, Lutron Electronics Co., Coopersburg, Pa.). Light intensity is controlled by providing 0– to 120–V signal to each dimmer. The voltage signals originate from a dedicated controller with four separate outputs (4 zones; Model GRX–3104, Lutron Electronics Co.). An additional module allows intensity to be programmed on a time basis or in relation to dawn and dusk calculated on the basis of geographic location, date, and time (Model GRX–RS232, Lutron Electronics Co.). The initial setup and programming requires the use of a computer and a serial connection. Programming can be done manually; however, an easy–to–use interface program that runs under Microsoft Windows 9x is available at no cost. Lighting control is associated with scenes. Each scene has a defined start and end time, and intensity. The transition between consecutive scenes is also programmable. The controller itself can accommodate four scenes; two additional modules allow for a total of 12 scenes (Model NTGRX–4S, Lutron Electronics Co.). Normally, outputs for zones one and two are used to control the two variable dimmers. An auxiliary relay with a spring–wound 30–min timer allows the inputs to the dimmers to be temporarily switched to zones 3 and 4. This relay allows the normal light intensity to be switched on a short–term basis to a programmed alternative, time dependent, intensity. In addition, the alternative intensity can be adjusted in real time. For example, when animals are in the chamber, lights are normally off or very dim during the night. In the event of an emergency, the light intensity can be temporarily increased by turning the timer. The default increase can be just enough light to see. If more light is needed to address the problem, the intensity can be increased manually. In any case, lighting will revert to the normal schedule once the timer resets.

Alarms

The system controller activates all alarms with one exception. The smoke detector in the supply duct has a contact closure monitored by the controller; however, it is also hardwired to shut down the chiller, fans, and electric heater if smoke is detected. The controller can activate a 90–dB horn mounted on the chamber, a 130–dB horn mounted on the building housing the chamber, and an automated telephone dialer. The dialer allows a unique message and dialing strategy to be associated with each of two contact closure inputs. One contact is used for emergencies requiring immediate intervention; i.e., something that endangers the health of the animals. Conditions that require immediate intervention are smoke detection, no air flow for over 3 min, or excessive chamber temperature. The second contact is used for warnings. Conditions that result in a warning include failure of cooling or heating equipment, or chamber temperature, or air flow out of accepted bounds. To prevent repeated callings by the telephone dialer when an alarm condition exists, the telephone dialer is enabled at the onset of an alarm and then every 30 min until the problem is resolved. Furthermore, for non–emergency warnings, audible alarms and the telephone dialer are disabled overnight. Opening either door to the chamber disables audible alarms and the telephone dialer.

Measured Variables

Temperature and Humidity

Measurements of air temperatures outside, in the supply duct after the electric heater (fig. 1), and in the enclosure housing the differential pressure transducers (discussed below) are made using 100–Ohm RTD sensors with 4– to 20–mA outputs (Model 533–376, Siemens Building Technology). These thermocouples are theoretically accurate to within 0.7°C. No attempt was made to validate this accuracy as outside temperature is used only to determine whether to enable the chiller or electric heater, and temperature measurements from the sensor after the electric heater are used only in intermediate control loops where reproducibility, but not accuacry, is required.

Temperature and relative humidity of air discharged into and leaving the chamber (fig. 1) are measured using duct mounted transmitters with 4– to 20–mA outputs (Model HMD60U, Vaisala, Woburn, Mass.). Accuracies are maintained to within of 0.6° C and 3% relative humidity using a calibration instrument and probe designed to work with the transmitters (Model HM141, Vaisala). The tip of the probe is threaded through a small hole in the duct adjacent to a transmitter. The instrument is connected to an electric port on the transmitter and the transmitter is then adjusted. The instrument and probe are laboratory calibrated using a National Institute of Standards and Technology traceable thermometer and a humidity calibration instrument consisting of a housing with two wells containing salt solutions which produce known vapor pressures (Model HMK15, Vaisala).

Air Flow

The air flow station is 46 cm (18 in.) in diameter (AMS–853–18, Ultratech Industries, Garner, N.C.). Static pressure is measured using stainless steel pitot tubes around the perimeter of the station. Cross–sectional pitot tubes are used to measure the average total pressure. The air velocity calculation (below) is based on the assumption that air flow is laminar. To facilitate laminar flow through the station, the station is located within a straight segment of duct, with seven lengths of the duct diameter before and three after the station. In addition, the station incorporates an egg–crate structure at its entrance with vanes separated by 1.9 cm (.75 in.). Thick–walled polyvinyl tubing is used to connect total and static pressure ports to two differential pressure transmitters with full scale accuracies of 0.25% (Ashcroft XLDP, Dresser Industries Inc., Stratford, Conn.). The full–scale ranges of the transmitters, 0 to 0.25 and 0 to 1.25 mb [millibar; 0 to 0.1 and 0 to 0.5 cm of water (cw)], correspond to 0– to 10–mA outputs. The static pressure port is also connected to a barometric pressure transmitter with a full scale accuracy of 0.03% and a 0– to 5–V output (Model PTB100A, Vaisala). The pressure transmitters are housed in a temperature controlled enclosure (40.6 × 30.5 × 20.3 cm, 21.1 ± 1.7°C, National Electrical Manufacturers Association Type 4, Electrographics Intern. Corp., Warminster, Pa.) located adjacent to the station. Air temperature and humidity measurements from the exhaust duct are the remaining measurements used in air flow calculations.

Calculated Variables

Most of the measured variables are used in control algorithms without transformation. However, there are a few variables that are derived; air density, air velocity, mass air flow, the setpoint for the temperature of air discharged into the chamber, and the cold air discharge setpoint.

Air Density

Air density (D; kg/m^3) is calculated using the Ideal Gas Law. However, water vapor in the air makes the air less dense. To account for this difference, it is possible to calculate a virtual temperature (Tv; °K); i.e., the temperature that dry air would have if its pressure and specific volume were equal to those of a given sample of moist air. To calculate the virtual temperature, first the saturation vapor pressure (Es; mb) is determined, and then the actual vapor pressure (E; mb). The virtual temperature is then used to calculate air density (USATODAY, 2000). The measured variables used in the calculations are barometric pressure (P; mb), relative humidity (RH; 0 to 1), and temperature (Tc; °C).

$$Es = 6.11 \times 10.0\hat{}\,(7.5 \times Tc /(237.7 + Tc))$$
$$E = RH \times Es$$
$$Tv = (Tc + 273.2)/[1 - (E/P) \times (1 - 0.622)]$$
$$D = 100 \times P/(Tv \times 287)$$

Air Velocity

Air velocity (V; m/min) is calculated as a function of the velocity pressure (Pv; mb) measured using the air flow station and the air density as calculated above. Measurements from the low range differential pressure transmitter are used until measurements exceed the maximum range (0.25 mb) of this transmitter and then measurements from the higher range transmitter are used. The equation used in actual practice is in U.S. customary units; the metric equivalent was derived from this equation (Dwyer Instruments, 1997). The estimated volumetric air flow (Mm; m^3/min) is calculated by multiplying the cross–sectional area of the air flow station times the calculated air velocity.

$$V = 848.2 \times (Pv/D)^{1/2}$$

$$Mm = 0.1642 \times V$$

In practice, velocity estimates are often based solely on differential pressure measurements; however, use of nominal values for air temperature, relative humidity, and atmospheric pressure in calculations instead of actual measurements can result in significant errors (table 1). The factor with the least impact on velocity calculations is humidity. Still, errors in estimating velocity if relative humidity is ignored can exceed 1% at high levels of humidity.

Mass Air Flow

The system controller is programmed to control mass air flow. Mass air flow is the number of molecules actually passing through the air flow station over time. When a volumetric air flow (Ms; m^3/min) is entered as a setpoint, the actual air flow setpoint (Ma; m^3/min) is adjusted so that the mass flow under current operating conditions is equivalent to the mass flow of the entered volumetric air flow at 1014 mb and 21.1°C (70°F). Based on the Ideal Gas Law:

$$Ma = Ms \times ((273.2+Tc)/(273.2 + 21.11)) \times (1014/P).$$

Chamber Discharge Setpoint

One goal of the control system is to maintain the measured exhaust temperature at a desired setpoint. The simplest control structure that could potentially accomplish this task would be to control the position of the steam valve based on the measured exhaust temperature. Unfortunately, this control structure is unstable because of the high gain associated with the steam valve and the long time constant, due to the distribution volume of the chamber, associated with the resulting changes in exhaust temperature. Instead, a desired air discharge temperature measured just after the steam coil is predicted based on the exhaust temperature setpoint and the measured exhaust temperature. The position of the steam valve is then controlled as a function of the desired and the measured discharge temperatures. The actual control algorithm is discussed below in the section on temperature control.

Cold Air Discharge Setpoint

Similarly, the position of the cold water valve is also controlled based on the air temperature measured just after the electric heater and a calculated cold air discharge set point. The actual control algorithm is discussed below in the section on humidity control.

ERROR MEASURING AIR VELOCITY

Computing the error associated with air velocity estimates, and hence volumetric air flow estimates, is a complex problem. One method for evaluating the maximum potential error is to use the error specifications for all components used in the velocity calculations and to construct the worst case scenario. The principle source of error is the differential pressure transmitters. The accuracy of the transmitters is specified as ±0.25% of full scale, which translates to an error of 0.00062 and 0.00311 mb for the 0.25– and 1.25–mb transmitters, respectively. In actuality, certificates for the actual transmitters showed a worst case accuracy of 0.15% so using the specified accuracy of 0.25% in calculations should result in significant overestimation of potential errors. A second source of error associated with the transmitters is in the zero and span. This source of error is dependent on the temperature of the transmitters. To reduce this error, the transmitters are housed in a temperature controlled enclosure. A third potential source of error is rounding error due to the 12–bit resolution of analog to digital conversions. However, the noise level for single measurements spans multiple bits. Assuming a zero mean noise distribution, the mean of repeated measurements is then essentially independent of the analog to digital sampling resolution. Thus, there is no need to consider sampling resolution in the analysis of overall error. As an example, this methodology was used to calculate the errors associated with the transmitters at the lowest air flow at which the chamber was designed to operate (23 m^3/min; table 2). The errors are about 1% and 5% for the 0.25 and the 1.25 mb transmitters, respectively. Similarly, this methodology can be used to account for the maximum potential errors associated with the measurements of the exhaust temperature and relative humidity, and barometric pressure. Figure 3 shows this calculated total potential error as a function of air flow. This figure clearly demonstrates why the use of two differential transmitters is required to keep the measurement errors below about 1%.

CONTROL FUNCTIONS

The primary control mechanism available with the digital controller is a PID (Proportional, Integral, Differential) feedback loop control subroutine. This subroutine is the basis for most of algorithms used to control chamber functions. The inputs for the subroutine are a measured or estimated control variable (Cv) and the associated setpoint (Sp), and

Table 1. Example of the impact on air velocity calculations if actual values are substituted for nominal values.[a]

Variable Differing from Nominal Value	Differential Pressure (mb)	Air Temperature (°C)	Relative Humidity (0 to 100)	Atmospheric Pressure (mb)	Calculated Velocity (m/min)	Difference (%)
None	0.040	18	40	1014	154.3	
Air temperature	0.040	37	40	1014	159.7	3.5
Relative humidity	0.040	18	90	1014	154.5	0.2
Atmospheric pressure	0.040	18	40	974	157.4	2.0
All	0.040	37	90	974	163.9	6.3

[a] The nominal conditions are 18°C, 40% relative humidity, and 1014 mb atmospheric pressure. The actual conditions are 37°C, 90% relative humidity, and 974 mb atmospheric pressure. Substitutions of actual values for nominal values of air temperature, relative humidity, or atmospheric pressure are made singularly and jointly.

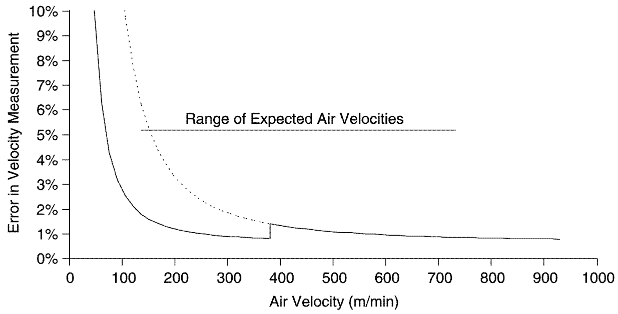

Figure 3. Estimates of the total error in air velocity measurements as a function of air velocity at conditions of 18.33°C, 50% relative humidity, and 1014–mb atmospheric pressure. Error estimates account for errors in measuring differential pressure, atmospheric pressure, relative humidity, and air temperature. Error estimates include an added 0.5% error to account for potential errors resulting from interactions of air flow with components of the air flow station. The transition in the solid line occurs at the point where the 0.25–mb transmitter fails. The dotted line shows error estimates if the 1.25–mb transmitter was used to make all differential pressure measurements.

the output is a process control variable (Pv). The other parameters in the subroutine are the proportional (Pg), derivative (Dg), and integral (Ig) gains as well as the sampling time (St). The shorthand notation for the subroutine is:

LOOP(Pv, Cv, Sp, Pg, Ig, Dg, St)

Air Flow

Mass air flow is controlled using the PID subroutine. The estimated volumetric air flow (Mm) is compared to desired mass air flow setpoint (Ma), and the output is a control voltage for the exhaust fan [Fv; 0–to–10V; LOOP(Fv, Mm, Ma, 100, 5, 0, 1)]. For stability, air density and velocity pressure measurements are low–pass filtered prior to calculating Mm. As the digital controller lacks provision for sophisticated filtering algorithms, a simple infinite impulse response filter is used where 30% of the current estimate is added to 70% of the previous value. Only the speed of the exhaust fan is actively controlled. The voltage signal to the supply fan controller is set to Fv minus 1.25 V. This difference results in a small negative pressure in the chamber (nominally 1~2 mm water measured using a large diameter clear tube which transverses the chamber). The PID subroutine is bypassed, and Fv is locked to its then current value, whenever a chamber door is opened and for 1 min after all doors are closed.

Main Cooling Coil

When cooling is needed, a PID subroutine is used to control the position of the cold–water valve. The cold air discharge temperature (Cd; °C) is compared to the cold air discharge setpoint (Cs; °C), and the output is a control voltage for the cold water valve (Cv; 0–to–10V; LOOP (Cv, Cd, Cs, 300, 3, 1000, 1). This setpoint is set to 2.2°C below the chamber discharge setpoint or the setpoint from the humidity control algorithm, whichever is lower. If the cold air discharge setpoint is below outside ambient temperature, cooling is deemed to be needed and control of the cooling valve is enabled. Once control of the cooling valve is enabled, ambient outside temperature must fall below the cold air discharge setpoint for 10 min before cooling is disabled and the cold–water valve is closed.

Table 2. Calculation of the maximum potential contribution of the differential pressure transmitters to the error in measuring air velocity.[a]

Transmitter Specifications:			
Range	0.2488	1.244	mb
Accuracy full scale	0.25%	0.25%	full scale
Span error	0.0056%	0.0056%	per °C
Zero error	0.0056%	0.0056%	per °C
Transmitter Environment:			
Maximum deviation from 21.11°C	1.67	1.67	°C
Chamber Specifications:			
Desired air flow	23.0	23.0	m^3/min
Air temperature	21.0	21.0	°C
Relative humidity	0	0	0 to 100
Atmospheric pressure	1014	1014	mb
Calculate:			
Air velocity = air flow / cross–sectional area	140.2	140.2	m/min
Duct diameter	0.457	0.457	m
Duct cross–sectional area	0.164	0.164	m^2
Reverse Calculate:			
Differential pressure	0.03266	0.03266	mb
Calculate:			
Air velocity (based on differential pressure)	140.2	140.2	m/min
Errors Due to Transmitter:			
Accuracy full scale	0.00062	0.00311	mb
Span and zero	0.00002	0.00012	mb
Total error due transmitter	0.00065	0.00323	mb
Net Error Due to Transmitter:			
Differential pressure + error due transmitter	0.03331	0.03589	mb
Calculate:			
Air velocity	141.6	147.0	m/min
% error	0.99%	4.83%	
Differential pressure – error due transmitter	0.03201	0.02943	mb
Calculate:			
Air velocity	138.8	133.1	m/min
% error	–0.99%	–5.07%	

[a] The calculations are based on the error specifications of the transmitters. The calculations are based on the worst–case scenario where all errors are assumed to be independent and additive.

Chamber Temperature

Temperature in the chamber is controlled primarily by controlling the valve regulating steam flow through the heating coil. In cold weather, the electric heater is used to heat the incoming air prior to the air reaching the steam coil. In warmer weather, cooling is controlled by controlling the position of the cold water valve to the main cooling coil. The cold air discharge temperature is regulated to provide sufficient cooling to allow control of chamber temperature using the steam heater and, under conditions where there is sufficient cooling capacity, to control humidity.

The first consideration in the temperature control algorithm is whether to use the electric heater. When outside ambient temperature falls to below 4°C, the first of the six stages of the electric heater is activated. For every additional 3.3°C decrease in outside ambient temperature, another stage is activated. To prevent cycling, stages are not deactivated until outside ambient temperature exceeds the associated activation temperatures by 0.3°C. As a failsafe, the electric heater is disabled if measured air flow falls below 20 m^3/min, the air temperature in the duct just downstream from the electric heater exceeds 15°C, or smoke is detected.

The next consideration is whether to enable use of the chiller. Use of the chiller is not enabled until the outside ambient temperature is above 8.3°C. To prevent cycling problems, a hysteresis function is used and the chiller, once enabled, is not disabled by a fall in ambient temperature until the temperature reaches 5.5°C. To save energy, the cold–water pump is turned on only when the chiller is enabled and the cold–water valve to the main cooling coil is not closed. The chiller is turned on only after a check of current flow to the cold–water pump confirms that the pump is working. Software timers are used to prevent rapid cycling of the chiller and the cold–water pump. After the cold–water valve has been closed for 5 min, the cold–water pump is turned off which disables and turns off the chiller. Once disabled, the chiller cannot be enabled for 5 min.

Animals housed in the chamber can be the source of a significant heat load. When animals are removed from the chamber for short periods for exercise, the abrupt change in heat load can affect the stability of the temperature control algorithms. To address this contingency, a switch is installed in the chamber the indicates whether the animals are in or out of the chamber. When the switch is moved to the out position, the chamber discharge temperature setpoint is locked to the then current value. When the switch indicates that the animals have been returned to the chamber, a software timer is started and 2 min later the discharge temperature setpoint is allowed to vary.

A simple PID–based control algorithm proved to be inadequate for controlling the chamber discharge temperature due to the characteristics of the steam valve, problems with steam pressure, and fluctuations in the cold air discharge temperature due to chiller cycling. The control algorithm is discussed in more detail in the Results and Discussion section.

Humidity

Humidity can be controlled only to the extent that the cold air discharge temperature can be lowered to the corresponding dew point for the exhaust temperature and humidity setpoints (Kuemmel, 1997). A PID subroutine is used to predict a cold air discharge setpoint based on the desired and measured exhaust humidity. The measured exhaust relative humidity (Hd; 0 to 1) is compared to the exhaust air relative humidity setpoint (Hs; 0 to 1), and the output is the humidity–based cold air discharge setpoint (Ch; °C; LOOP(Ch, Hd, Hs, 300, 6, 0, 1)). This setpoint is compared to the corresponding setpoint for temperature regulation, and the lower value is used as the cold air discharge setpoint.

Results and Discussion

Problems

Air Flow

The first problem encountered was in the measurement of volumetric air flow. The original specifications called for control of volumetric air flow to within 2% of the setpoint with a stated assumption that control to within 2% required that average air velocity be measured with a 1% accuracy. During construction, it was determined that the contractor did not understand this specification and had intended to provide a thermo–anemometer with a stated 1% accuracy. Unfortunately, dimensions of the original exhaust plenum were such that the air velocities through the plenum at low air exchange rates were below the threshold required for using the thermo–anemometer. Furthermore, due to the velocity profile across the plenum, the error bounds for using a single velocity measurement within the plenum to estimate volumetric air flow exceeds 10%. A more appropriate mass air flow measuring system was designed which resulted in the removal of a large segment of the exhaust duct and installation of the smaller round duct which contains the air flow station. One consequence of this substitution was an increase in the pressure drop across this segment. This increase reduced the maximum obtainable air flow rate from the design specification of 136 to 119 m^3/min.

During operation of the chamber, another problem related to measuring the volumetric air flow became evident. Over time, the voltage necessary to maintain a given air flow increased and the apparent maximum obtainable air flow decreased. It was determined that dust had clogged the pressure measurement ports in the air flow monitoring station. To remedy this problem, the static and total pressure ports are backflushed; an air gas cylinder with the regulator set at 210 kPa (30 psi) is connected to each port in turn for 30 s. Care must be taken not to expose the pressure transmitters to this high pressure. A good method to determine if the ports are clogged is to set the fans at maximum (10 V output to both supply and exhaust fans) and measure the resulting air flow. Any drop in measured air flow compared to the value obtained from an optimally functioning system can be assumed to be due to a blocked port. If backflushing the ports does not restore the measured air flow to the expected value, the fan belts should be checked.

Electric Heater

The original contract called for a six–stage electric heater. At start–up, it was determined that the heater had an interlock, based on air flow, that prevented operation of the heater unless air flow was sufficient to use all six elements. The result was that the heater could not be turned on except at the highest air flows. The assumption in the design specifications was that a fraction of the six elements could be used at lower air flows and that all elements could be used at the lowest designed air flow during periods of very low outside ambient temperature. The air flow interlock was bypassed, even though doing so voided the Underwriter's Laboratory (U.L.) listing, because the heater also contains a thermal interlock, and external measurements of air temperature, air flow, and the presence of the smoke provide the basis for additional failsafes.

Telephone Dialer

The telephone dialer initially installed did not allow the system controller to reset the telephone dialer; thus, the dialer would continue to place calls even after a problem was remedied. A new dialer that could be reset was installed.

Steam Control

There are three substantial problems related to the use of the steam valve to control the chamber air discharge temperature. First, responses to increased and decreased heating demands are not symmetrical. Opening the steam valve results in a very rapid increase in the air discharge temperature while there is a considerable delay in the fall of the discharge temperature after closing the valve. Second, the air temperature response to a change in valve position is complex and distinctly nonlinear; the mostly problematic response characteristics involve a large hystereis when the valve is less than 10–12% open and a transient response where discharge temperature actually increases for a short time after the valve is incrementally closed. Altering the control algorithms to keep the valve at least 15% open whenever possible reduced the impact of these two problems at the cost of increased energy consumption.

The final problem involves steam pressure. A central boiler serves all the buildings in the vicinity of the chamber. At each building, pressure from a main distribution line is reduced for local distribution. The tap for the steam valve and coil shares the same low pressure (80 KPa) distribution line with numerous high capacity heating units. Transient decreases in steam pressure at the control valve caused by simultaneous activation of a number of these heating units create a problem for the temperature control algorithm. If the gain of the valve response is increased to allow for compensation of these drops, the control system overreacts to normal fluctuations in chamber temperature. To address this problem, and to deal with the asymmetric temperature response to opening and closing the valve, the PID control algorithm for controlling air discharge temperature was modified to attempt to limit the rate of change of the valve position. This modification is also directed at alleviating problems with chiller cycling and is discussed in more detail below.

Cold Water Discharge Temperature

The automatic cycling of the chiller has a pronounced effect on the cold–water discharge temperature from the chiller (fig. 4). This cycling, and the resulting rapid increase in air temperature leaving the cooling coil, impacts the ability to regulate the exhaust air temperature (fig. 5). The temperature of the steam coil cannot be reduced fast enough to prevent a transient increase in exhaust temperature. This increase in exhaust temperature creates an error which, in turn, results in a decrease in the air discharge temperature setpoint. Thus, the apparent oscillations in the chamber discharge temperature setpoint are due to chiller cycling.

The effects of chiller cycling are mitigated when sufficient chiller capacity exists to allow regulation of the cold air discharge temperature. The cold air discharge control algorithm seeks to maintain a set cold air discharge temperature from the cooling coil. Thus, the algorithm damps the impact of chiller cycling on the cold air discharge temperature.

Humidity Control

The minimum cold–water discharge temperature from the chiller was initially set at 5.5°C, which, due to the efficiency of the single cold water coil, resulted in a minimum

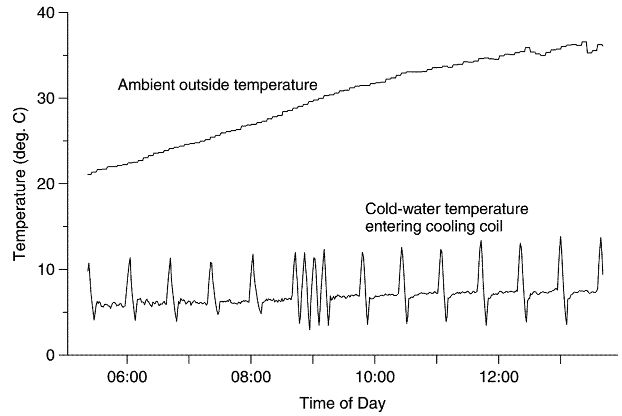

Figure 4. The upper line shows outside ambient temperature over time. The lower line shows the cold–water discharge temperature from the chiller. Note the cycling of the discharge temperature even at high ambient temperatures when the chiller is essentially running at maximum capacity.

obtainable dew point of approximately 11°C. At an exhaust temperature of 18.3°C, this dew point corresponds to a theoretical relative humidity of about 65%. In practice, due to the influence of the animals, the relative humidity when the dew point of supply air was around 11°C and the exhaust temperature was set at 18.3°C ranged from 68 to 75%. To allow for better humidity control, the nominal discharge temperature from the chiller was lowered to 2°C. This change reduced the maximum chamber relative humidity under the above conditions to around 55 to 62%. Addition of a second cold water coil to improve the cooling efficiency should theoretically reduce the maximum chamber relative humidity under the above conditions to around 50%, or to a theoretical dew point of 4.5°C which corresponds to a relative humidity of 40% at 18.3°C. Further reductions using cooling to remove moisture from the air are not practical as reducing the chiller discharge temperature below freezing can result in the coils freezing and blocking air flow. Absolute humidity control requires the use of renewable absorbents to remove moisture from the air.

Cooling Capacity

To reduce the initial cost of the system, the specified cooling capacity was limited to a 20–ton chiller with a setpoint of 5.5°C. A 30–ton chiller would have cost an additional $15,000. In operation, this capacity allows the chamber to be operated at 18.3°C with the lights on and six dry cows in the chamber when outside ambient temperature remains below about 27°C; or at 23.9°C when ambient temperature remains below about 35°C. The exact temperature differentials depend on the moisture content of the outside ambient air. In addition, as the capacity of the chiller is reached, the cold water discharge temperature from the chiller increases as a function of load. Thus, the minimum obtainable dew point rises which, in turn, places limits on humidity control. For long term studies, the operating range of the chamber could theoretically be extended by shifting the chamber schedule so that the chamber was dark during the day when ambient temperatures were highest. This lighting shift equates to an increase in cooling capacity of about 1 ton. For example, lights could be on 14 h per day from 1800 to

0800 h and, if necessary, cows could be milked at 1800 and 0700 h.

Auxiliary Cooling

To improve the efficiency and capacity for cooling the chamber, an auxiliary air handler was installed in the chamber so that air inside the chamber could be recirculated across an auxiliary cooling coil. Subsequent to the construction of the chamber, it was realized that significant problems existed relative to the use of the auxiliary cooling system. First, the aluminum cooling coil was not compatible with the ammonia in the air resulting from animal excretions. Second, for ammonia studies, estimating the quantity of ammonia released within the chamber would be greatly complicated by having to measure ammonia in the condensate from the auxiliary cooling system. The system was removed from the chamber. The economic loss due to purchasing, installing, and removing the system exceeded $10,000.

Operating Characteristics

Control algorithms for setting the chamber discharge temperature setpoint and for controlling the steam valve are outlined below. A number of alternative algorithms have been investigated and optimized for seasonal operating conditions and, more specifically, for dealing with the problems related to chiller cycling and to transient changes in steam pressure. However, these algorithms are not discussed in detail as many of the problems addressed by the algorithms are site specific. Instead, the control algorithms will be outlined in sufficient detail to allow users to develop their own site–specific algorithms.

The basis of the algorithm used to control the steam valve is to establish a temperature setpoint for the air discharged into the chamber; to generate a nominal output control voltage for the steam valve based on this setpoint and the measured discharge temperature; and to adjust the nominal control voltage based on dynamic changes in the discharge temperature. If the measured discharge temperature is close to the setpoint (e.g., within 0.2°C) no adjustment is made and the nominal voltage is output to the valve actuator. Otherwise, the nominal voltage is adjusted depending on whether the measured discharge temperature is above or below the setpoint temperature, the rate of change of the measured discharge temperature, and whether the temperature is rising or falling. Additionally, the measured and desired exhaust temperatures can be used as a basis for adjustments.

The first step in the control scheme is to calculate the chamber discharge temperature setpoint (Ds; °C) based on the measured chamber exhaust temperature (Tm; °C) and the exhaust temperature setpoint (Ts; °C; LOOP (Ds, Tm, Ts, 300, 10, 1000, 10)) at 10–s intervals. The second step is to generate an intermediate control voltage (Ia; 0–to–10 V) for the steam valve based on the measured chamber discharge temperature (Dm; °C) and the discharge temperature setpoint (LOOP (Ia, Dm, Ds, 0, 5, 0, 1)). The nominal output control voltage (Ib; 0 to 10 V) is calculated by averaging the intermediate control voltage over the prior 60 s. Every 5 s, to prevent excessive movement of the steam valve, the actual voltage output to the steam value is updated as follows:

IF Dm is less than Ds THEN GO TO *Branch0*
IF Dm less than Ds + 0.2 THEN output Ib
ELSEIF Dm less than the previous Dm – 0.02 THEN output Ib (Dm falling)
ELSEIF Dm less than Dm + 0.7 THEN output Ib – 0.7V
ELSE output Ib – 1.5V
EXIT SUBROUTINE
Branch0:
IF Dm greater than Ds – 0.2 THEN output Ib
ELSEIF Dm greater than the previous Dm + 0.02 THEN output Ib (Dm rising)
ELSEIF Dm greater than Dm – 0.7 THEN output Ib + 0.3V
ELSE output Ib + 1.0V

With this control algorithm, the exhaust temperature from the chamber with six pregnant heifers in the chamber, no chiller cycling, and no transient changes in steam pressure, can be maintained within 0.06°C of the set–point. With chiller cycling or transient changes in steam pressure, the error is normally less than 0.11°C with very short duration deviations of up to 0.17°C (fig. 5). Following a transient change in steam pressure which occurred during a episode of chiller cycling, the exhaust temperature deviated from the set–point by 0.28°C and did not return to the setpoint for 15 min. Fortunately, the simultaneous occurrence of both problems is rare as the chiller is seldom used when ambient temperatures are low enough to require use of significant steam resources for heating the building. The chamber was also tested with a space heater (1500 W) instead of animals in the chamber. Control tolerances were similar.

The temperature control algorithm has the potential ability to follow a time–varying setpoint such as a diurnal temperature pattern. This ability has not been tested in detail; however, the response to a step change in exhaust temperature setpoint indicates that following a time–varying setpoint should not be a problem (fig. 6). Tests with an empty chamber, with the lights on, indicate that it may be possible to eliminate the overshoot by limiting the rate of change of the temperature setpoint.

The ability to control relative humidity in the chamber is a function of the operating temperature of the chamber, the minimum obtainable dew point temperature, and moisture

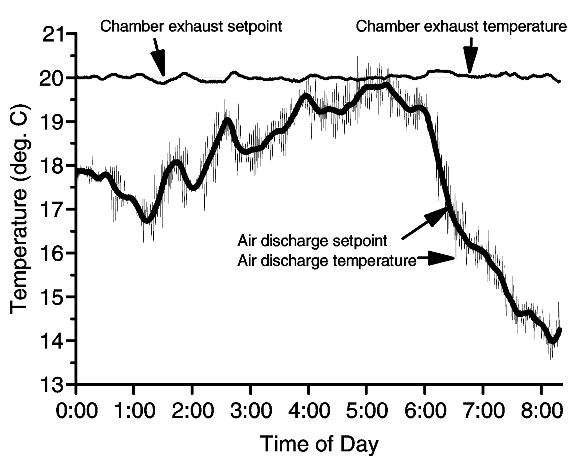

Figure 5. The upper line shows the measured exhaust air temperature over time. The temperature setpoint is 20°C. The lower, bold, line shows the air discharge setpoints over time surrounded by actual discharge temperatures. The oscillations in the discharge temperature setpoint are due to the periodic fluctuations in the cold water discharge temperature resulting from chiller cycling.

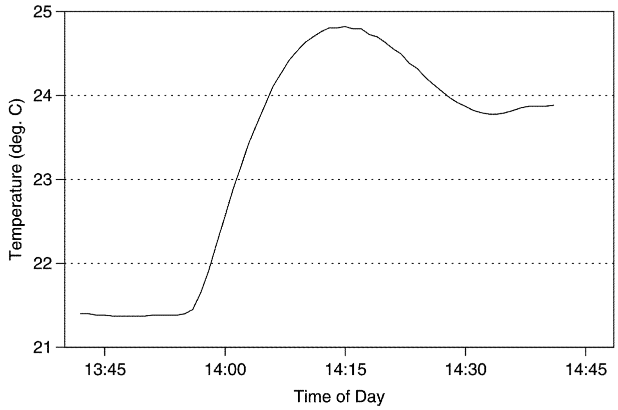

Figure 6. Exhaust air temperature response when the setpoint was changed from 21.39 to 23.89°C. Air flow was 12.5 air exchanges per hour.

production by animals in the chamber. If the chamber is operated at 18.3°C (65°F), which is the optimal temperature for lactating dairy cows, the minimum obtainable dew point temperature of 2°C allows for only limited humidity control. In practice, with six dry cows in the chamber, relative humidity was generally below 60% but went as high as 80% during periods of high ambient outside temperature. This range in humidity is acceptable for studies of ammonia release as the animals are not stressed by the humidity and gas transfer rates are not affected by humidity. However, it was noted that during the winter when relative humidity levels in the chamber were less than 20% due to extremely cold outside ambient conditions, that ammonia emission rates were reduced. Presumably, the low humidity levels allow for rapid crust formation on manure, which, in turn, reduces ammonia volatilization (personnal observation). If the operating temperature of the chamber is raised to 26.7°C (80°F), the ability to limit the maximum chamber humidity level is greatly increased under most ambient outside conditions, and there is sufficient cooling capacity to allow the chamber temperature to be maintained at the temperature setpoint for all but a few very hot days each year.

Estimated air velocities initially showed a small oscillation. It proved difficult to characterize the oscillation because the controller does not support time–based sampling rates faster than one sample per min. Data can be displayed at approximately 1–s intervals, but not with an accurate time–base. Consideration was given to using an external monitoring system to examine the oscillation; however, external monitoring proved unnecessary as the oscillation was dealt with by using infinite impulse response filters for both the calculated air density and the measured differential pressure. Filtering allows measured air flow to be controlled to within 1.5% of the setpoint. When measurements of air flow are averaged over one minute intervals, the averages essentially do not differ from the setpoint. Thus, in relation to processes involved in gas emission rates, the error in controlling volumetric air flow is negligible. With negligible control error, the error in volumetric air flow is essentially the measurement error. With judicious choice of operational air flows, this error can be kept below 1% (fig. 3). There was one exception to the above. On a day with very strong gusty winds, measured air flows deviated from the setpoint by 2 to 3% during and for 10 s following each gust.

The only independent verification of the accuracy of the volumetric air flow calculations involved recoveries of ammonia released in the chamber (Lefcourt, 2001). These recoveries indicated that the average air flow did not significantly differ from setpoints. However, recovery studies are not as accurate as the theoretical flow measurements and are only indicative of the average flow rate. The accuracy of the volumetric air flow calculations in the time–frame used to control the fans could not be checked because there is no alternative accurate and cost effective method for measuring the instantaneous volumetric air flow.

CONSIDERATIONS

Cooling

Incorporation of a 2000–L reservoir between the chiller and the cooling coil should eliminate the effects of chiller cycling and provide a buffer for peaks in ambient temperature. The reservoir should also reduce cooling costs by allowing the chiller to be cycled more efficiently. By using a surplus water tank as the basis of the reservoir, the estimated cost of adding the reservoir to the existing cooling system is less than $2,000. The major cost is for propylene–glycol antifreeze. Addition of a second coiling coil downstream from the main cooling coil along with a second chiller, or a higher capacity chiller, to increase cooling capacity should improve humidity control and increase the operating range of the chamber.

Steam Pressure

Adding an independent pressure controller between the high pressure steam line serving the building and the steam valve would eliminate the effects of transient changes in steam pressure due to heating units sharing the same low pressure steam line.

Lighting

Diurnal lighting patterns can influence biological processes (Lumsden and Millar, 1998; Turek and Van Reeth, 1996). The lighting system was designed to allow simulation of the range of diurnal lighting patterns that can result from ambient solar radiation. If such control is deemed unnecessary, a simple time switch can be used for lighting control and fluorescent bulbs can be used instead of incandescent bulbs. The use of fluorescent lighting would reduce the heat load due to lighting from 2945 W to about 650 W.

The existing lighting control system can be cumbersome to use. It works very well for simple transitions; however, it is not easy to program a diurnal lighting pattern to simulate natural lighting. An alternative would be to use the digital system controller to control lighting. For example, a 0– to 10–V output could be used to directly control dimmers with analog input capabilities. As an added benefit, using this alternative it would be possible to measure and actively control light intensity in the chamber. If the decision is made to use an independent lighting controller, it would be possible to reduce costs by using a single zone controller instead of a four zone controller. The one zone could be used to control all dimmers. The lighting override signal could be generated using a voltage divider or by using 120 VAC as the control voltage.

Fans

Fine control of the supply fan is not necessary for control of volumetric air flow. In fact, setting the control signal to the supply fan to a fixed value can reduce the error in controlling air flow at the expense of allowing the differential pressure between the chamber inside and outside to vary. If this is an acceptable alternative, a less sophisticated and less costly fan controller can be used for the supply fan.

Digital System Controller

The digital system controller used in this installation was designed primarily as a building controller. Its primary advantage is that it is an industrial unit with many failsafes including the ability to alter program code in real time without effecting the operation of the chamber. There are three significant disadvantages with this type of controller. First, the control structure is optimized for use of PID subroutines and there is no provision for other types of control techniques; e.g., a control system based on modeling the system. Second, it is impossible to implement sophisticated software filters as sampling methodology and rates are inadequate. Third, it is difficult to program the device. Program lines have to be erased and re–written one line at a time. Program variables (points), which have complex attributes, cannot be edited but have to be completely redefined. An extra–cost management program is available that will alleviate some of these programming problems. Some specific advantages are that it is easy to set data acquisition parameters, to view user selected operating variables in real–time, to view acquired data, and to do all this using a modem. Data are displayed as scrolled text. If this controller is selected for use, it is recommended that the system be purchased with a 40– instead of a 26–module backplane.

An alternative to using this digital controller would be to use a computer–based control system. This alternative would probably save money and allow more versatility. However, care must be taken to assure the system is environmentally robust and includes adequate failsafes.

Temperature Control Box

Thermal relays are used to control temperature within the enclosure housing the differential transmitters. Because of the inherent inaccuracy of these relays, it has proved difficult to reliably control the temperature within the specified bounds. As the temperature within the enclosure is already monitored by the digital controller, using the controller to regulate the enclosure temperature would require only the substitution of contact closure outputs for the two relays controlling heating and cooling, and a minor addition to the control program.

CONCLUSIONS

Advances in available technology allow for the construction of large environmental chambers with very tight control tolerances at reasonable cost. The described chamber, constructed from a modular cold–box using off–the–shelf components, allows control of chamber temperature to within 0.17°C of the setpoint. The sources of most of the error in regulating temperature are spikes in the cold air discharge temperature due to chiller cycling and transient changes in steam pressure. Suggested are methods to alleviate these problems if tighter control of chamber temperature is desired. The chamber design also allows exhaust air flow to be measured and controlled to within 1% of the setpoint. These guidelines for designing and operating a large environmental chamber should help avoid some potential pitfalls and help reduce the cost of a new chamber.

REFERENCES

Andersson, M. 1995. Ammonia volatilization from cow and pig manure. Results of laboratory studies with a new climate chamber technique. Report 98, Department of Agricultural Biosystems and Technology, Swedish Univ. Agricultural Sciences, Lund.

Brown–Brandl, T. M., M. M. Beck, D. D. Schultze, A. M. Parkhurst, and J. A. DeShazer. 1998. Modification, operation, and error analysis for an indirect calorimeter for within–chamber control and data acquisition. *Transactions of the ASAE* 41(4): 1115–1118.

Dwyer Instruments. 1997. Calculating air velocity from velocity pressure. In *1998 Controls and Gages Catalog*, 33F. Michigan City, Ind.: Dwyer Instruments Inc.

Elzing, A., and G. J. Monteny. 1997. Ammonia emissions in a scale model of a dairy–cow house. *Transactions of the ASAE* 40(3): 713–720.

Kuemmel, B. 1997. Temp, Humidity & Dew Point ONA. http://www.bookcase.com/library/ faq/archive/meteorology/temp–dewpoint.html.

Langhans, R. W., and T. W. Tibbitts. 1997. *Plant Growth Chamber Handbook*. Iowa Agriculture and Home Economics Experiment Station Special Report No. 99. Ames, Iowa: Iowa State Univ.

Lefcourt, A. M. 2001. Large environmental chamber: calibration of ammonia release. *Applied Engineering in Agriculture* 17(5): 683–689.

Lumsden, P. J., and A. J. Millar (eds.). 1998. *Biological Rhythms and Photoperiodism in Plants.* Washington, D.C.: Bios Scientific Publishers.

Monteny, G. J., D. D. Schulte, A. Elzing, and E. J. J. Lamaker. 1998. A conceptual mechanistic model for the ammonia emissions from free stall cubicle dairy houses. *Transactions of the ASAE* 41(1): 193–201.

Nienaber, J. A., and A. L. Maddy. 1985. Temperature controlled multichamber indirect calorimeter–design and operation. *Transactions of the ASAE* 28(2): 555–560.

Turek, F. W, and O. Van Reeth. 1996. Circadian rhythms. In *Environmental Physiology, Vol. 2*, eds. M. J. Fregly and C. M. Blatteis, 1329–1359. New York: Published for the American Physiological Society by Oxford University.

USATODAY. 2000. Virtual temperature and humidity. http://www.usatoday.com/weather/ wdencalc.htm.